Reprintad From
PVP — Vol. 169 Design and Analysis of
Piping and Components
Editors: Q. N. Truong, E. C. Goodling, JR.
J. J. Balaschak, and G. E. O. Widera
Book No. H00484 — 1989

The American Society of
Mechanical Engineers

THE ART OF CHECKING PIPE STRESS COMPUTER PROGRAMS

Liang-Chuan Peng
Peng Engineering
Houston, Texas

ABSTRACT

With the computer getting more and more sophisticated, the chance of getting a bug in a program or a misapplication in an analysis also becomes more and more likely. Analysts need some rules of thumb to quickly spot problem areas and to make a quick check if necessary. This paper outlines some of the general rules used in checking boundary conditions, unbalanced forces, and irregularities. It also uses specific examples to demonstrate the checking of some elementary functions. Special discussions are given on advanced features such as support friction, thermal bowing, and expansion bellow elements.

INTRODUCTION

With the new requirements given on the design of a modern plant piping, the only practical tool for the design analysis is the computer. The computer program designed for pipe stress analysis gets more and more sophisticated every day. Some programs have gone through several generations of development employing completely different background of personnel. The new generation normally will not touch the good work done by their predecessors. Instead, they make layers of shells around the existing work. The completed program becomes very disorganized. Therefore, it is safe to say that a modern pipe stress computer program is bound to have some inconsistencies.

Pipe stress analysts are normally too timid in challenging a well established computer program. However, if we recognize that to err is computer program, we may be able to more objectively ensure the quality of our analysis. It is important to realize that everything has its so called norm. In other words, if something looks unrealistic then it probably is unreal. Therefore, it is important to be able to look at the output and point out the irregularities that might exist. That is the art. From time to time we have seen some experienced engineers who are able to judge whether a system is satisfactory just by looking at the model. The computer analysis is just a comfirming check. However, they are the exceptional rather than the normal.

The inconsistent results in an analysis comes either from the bug in the program or from the misapplication of the program. Nowadays, people like to boast that you don't even need to read the manual to use their computer program. The so called user friendly is probably what they intended to say, but somehow the impression they give is not. You type in some data, then you get some results. It sounds easy, but is scary. To ensure a good analysis the analyst has to have at least a clear picture of what the program functions are. He or she should also be able to spot the inconsistencies when they occur.

PROGRAM VERIFICATION

A program is systematically verified before being released for production. The verification involves almost every step of the program's operation and function. The results of the verification are documented in the verification reports. This is the function of the program developer and should not be a burden to the users.

Verification by the user is occasionally required by the inhouse QA procedure, or to simply satisfy the curiosity of the user or the boss. To an analyst, to be able to personally verify a couple of analyses will definitely increase his or her confidence in the program. The most common approach of the verification is to check against known results. The book by Kellogg Company [1] contains quite a few hand calculation results which can be checked against the expansion stress calculation. A more formal calculation intended to be a benchmark was published by ASME [2] in 1972. Unfortunately, this benchmark problem contains some misprints, which have never been corrected, and also the unusual non-circular cross section elements. Because of these difficulties, the problem has created a huge frustration in the piping industry. Everywhere, engineers are trying to make a comparison in vain. Later in 1980 U.S. NRC published a set of representative piping benchmark problems [3]. This set of problems was taken from real systems laidout in nuclear power plants. It is mainly used to check the earthquake analysis using the response spectra method.

The benchmark problems check only the general behaviors of the program. The general behavior of a given program differs very little from the original black box on which most of the programs are based. Therefore, very little deviation shall be expected from these tests. The most important items to be concerned with are the ones particular to individual programs. These items need to be checked very discretely.

DEVIATION

In comparing the test results with published or benchmark results, the relative deviation is used. The term error is not used because the difference might be caused by the error of the published or so called known results. Even the so called exact solution might have some seemingly insignificant terms ignored. However, if the deviation is small then there is a good chance that both the testing program and the benchmark are correct. This is more so when then testing program uses an entirely different solution technique than that used by the benchmark.

In evaluating the deviation, some common sense has to be applied to avoid unnecessary arguments. For given quantity, R, whose exact solution is shwon in Figure 1 (a). Its corresponding result, R', from the test program may be shifted to as shown in Figure 1 (b). Then by some methods of evaluation,

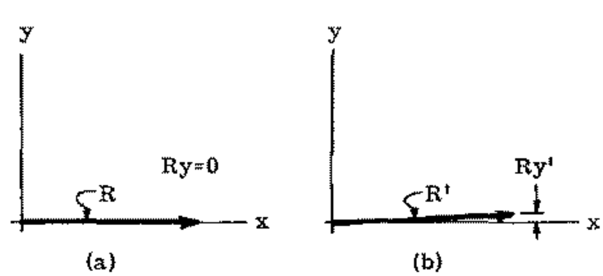


Figure 1, Standard Deviation

it may be concluded that there is no comparison at all. Because the deviation is essentially infinite on component Ry. But we all know that the real difference between the two solutions is very small. This can be easily proved because if we rotate the axes by 45 degrees, the deviation will almost disappear completely. The point is that a number is meaningless if its quantity is entirely dependent of the selection of the coordinate axes. Therefore, it is important to have the deviation properly defined as follows :

$$\text{dev}(Ry) = (Ry' - Ry) / Ry \qquad \text{(Meaningless)}$$

$$\text{dev}(Ry) = (Ry' - Ry) / R \qquad \text{(Local)}$$

$$\text{dev}(Ry) = (Ry' - Ry) / Ro \qquad \text{(Global)}$$

Where R is the resultant quantity at the point of interest, and Ro is the maximum resultant quantity in the entire system analyzed. The global deviation is introduced, because at a given point the resultant quantity itself may be insignificant. Whether it is significant or not, the tool to measure is the global comparison. The evaluation of the local deviation requires some personal judgement, but the global deviation should be limited to about 10 percent.

BOUNDARY CONDITIONS

The first step in quick checking an analysis is to make sure that the results match the boundary conditions of the systems. This can be done easily with the help of a good output arrangement. Most computer programs have a separate report for the anchor and support forces and moments as shown in Table 1. For this particular one [4] the friction and the pipe displacements are also given. This makes the checking of the boundary condition very easy.

By using reports such as Table 1, the boundary conditions can be checked directly by looking at the pipe displacements. At an anchor point the pipe displacement should be the same as the input displacement, and at the limit stop location the pipe displacement shall be equal to or smaller than the gap specified. However, it should be noted that the support displacement specified in the input is for the support structure. The actual pipe displacement at that point may or may not be the same as the support depending on the rigidity of the support. If the support is rigid then the pipe and the structure will have the same displacement. But if the support is flexible then the pipe displacement and the support movement are different as shown in Figure 2.

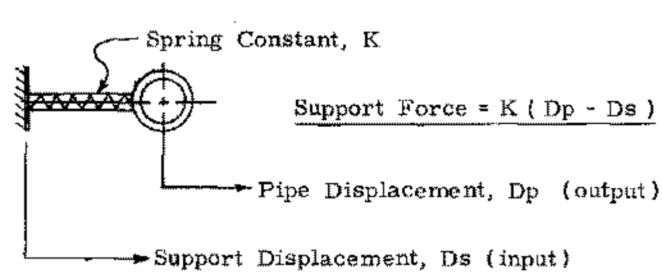


Figure 2, Support and Pipe Displacements le 1, Anchor and support Loads

88/12/12. PENG ENGINEERING - SIMFLEX-II (REL-4.0) (ANSI-B31.3) PAGE 11

SAMPLE PROBLEM, 1989 ASME/JSME PVP CONFERENCE
FRICTION FACTOR 0.4 ON ALL VERTICAL SUPPORTS

CASE 1 TH + WT RESULTS LOAD= THM,WGT,BNG,FOR,UFR,CSP,PRES
****

*** ANCHOR AND SUPPORT FORCES - INCLUDING FRICTION (ACTING ON SUPPORT) ***

		SUPPORT FORCE AND MOMENT						FRICTION				DEFLECTION			
SUPT	DATA	FORCES (N)			MOMENTS (N-M)			FORCES (N)			T(N-M)	(MM)			NOTES
TYPE	PT	FX	FY	FZ	MX	MY	MZ	FFX	FFY	FFZ	FMT	DX	DY	DZ	
VESS	5	-4552	2898	2491	-1951	-739	-763	0	0	0	0	-.0	-3.6	.0	
	LOCAL	2898	-4552	-2491	-739	-1951	764								
		RADIAL	MERIDN	TANGNTL	TORSION	CIRCUMF	LONGTDL								
SPRG	20	0	-6845	0	0	0	0	0	0	0	0	2.9	-6.1	-1.0	
LSX	65	0	0	0	0	0	0	0	0	0	0	3.0	-.0	34.4	INACTIVE
LSX	65	1401	0	0	0	0	0	0	0	0	0	3.0	-.0	34.4	
NLY	65	0	-10154	0	0	0	0	353	0	4047	0	3.0	-.0	34.4	
STX	70	1389	0	0	0	0	0	0	0	0	0	.0	-.0	16.4	
NLY	70	0	-5041	0	0	0	0	0	0	2017	0	.0	-.0	16.4	
LSX	90	-7646	0	0	0	0	0	0	0	0	0	-6.0	-.0	-1.7	
LSX	90	0	0	0	0	0	0	0	0	0	0	-6.0	-.0	-1.7	INACTIVE
NLY	90	0	-450	0	0	0	0	-173	0	-47	0	-6.0	-.0	-1.7	
LSZ	100	0	0	-7647	0	0	0	0	0	0	0	-7.7	1.8	-6.0	
SPRG	105	0	-2976	0	0	0	0	0	0	0	0	.1	6.8	4.8	
VESS	120	7043	-2298	-2871	-3087	-5748	-7368	0	0	0	0	.0	2.0	8.0	
	LOCAL	2299	-2871	-7043	5748	-7369	3088								
		RADIAL	MERIDN	TANGNTL	TORSION	CIRCUMF	LONGTDL								
SPRG	150	0	-1535	0	0	0	0	0	0	0	0	-3.2	29.5	40.4	
SPRG	160	0	-6599	0	0	0	0	0	0	0	0	11.4	17.1	.0	
STZ	160	0	0	2009	0	0	0	0	0	0	0	11.4	17.1	.0	
ANCH	180	2188	-371	4	-1267	3501	-9188	0	0	0	0	-2.0	1.5	.0	
NET FORCES		-177	-33371	-6014				180	0	6017	0				

For systems which include the support friction, both the force and the direction of the friction can be readily checked against the normal support force and the pipe movement.

SYSTEM EQUILIBRIUM

The pipe stress analysis result, regardless of the method used to get it, shall still conform the law of equilibrium. The summation of the forces and moments applied at a given point shall be zero, and the summation of the forces and moments applied to the entire system shall also be zero.

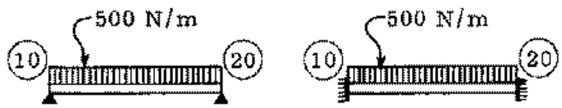
Needless to say that to check the equilibrium for every point in a system manually is not practical. But if there is any doubt about a given point, then it can be checked manually. In well designed programs, there is a scheme to automatically check and record the equilibrium of all the points in the system. The analyst should always look for messages to see if any significant unbalanced force has been detected. A significant unbalanced force always signals a problem in the analysis.

The total system equilibrium can be checked by using the support load table given in Table 1. In this table the total system forces are summarized at the bottom. In a system without any external forces entered explicitly, the vertical force should be the same as the total weight load. The horizontal force should be equal and in the opposite to the horizontal friction force. This is very fundamental, but can be missed by even the expert. For instance, in Problem No. 6 of the ASME 1972 verification book [2], one of the solutions presented has an apparent error in the Y-direction support force. This can be checked by the law of equilibrium but the writer preferred to have it explained as the difference of the programs used in the comparison.

ELEMENTARY FUNCTIONS

It is true that the pipe stress computer program is designed to handle an assembly of pipes. However, it should still be able to calculate some very simple situations. A pipe stress program consists of mainly two types of elements, straight pipe and curved pipe. If the program is to function properly then these two basic elements have to function properly. Therefore, if we can check out the basic function of these two elements, we will have more confidence in the program.

The straight pipe element is just a beam. Its function can be checked against the beam formula we have learned from text books. However, there are a few differences that need to be mentioned. One of them is that some program approaches are not as clear as the text book. Take the two uniformly loaded beams as shown in Figure 3 for example, if you run them through the computer you may find that there is no stress at all in one, or even both, of the cases. In (a), because the program evaluates only the stresses at node points 10 and 20, and the stresses at these two points happen to be zero. There is no reason that a program can not be programmed to find the maximum stress of the entire beam element, but this is not done normally. The reason is that a complete pipe stress analysis can involves several load cases. If all the maximum stresses at each element are to be combined together regardless of their location, then the calculation can become overly conservative. Also the simple beam condition does not really exist in a piping system. If required, an additional point at the midspan can be entered. The case (b), on the other hand is somewhat more troublesome. In some finite element programs the uniform load is divided into nodal loads which are applied at the node points. In the fixed beam case, the uniform load is divided into two concentrated loads which are applied at the ends. This will produce the proper reaction force, but no reaction moment nor beam stress. Some programs of this type are still widely used in the piping industry. Analysts should make themselves aware of the problem involved.



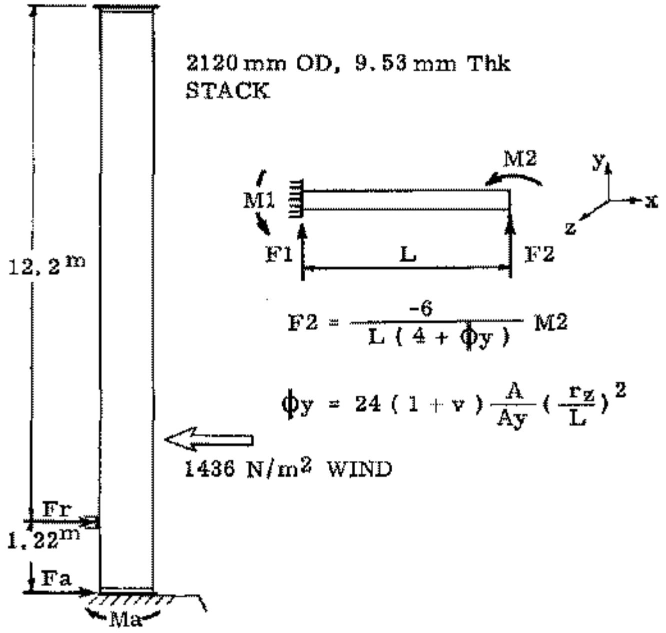
(a) Simple Beam (b) Fixed Beam

Figure 3, Beam Paradox

Another item that needs to be mentioned is shear deformation. The shear deformation is not normally included in the beam formula we use, but it is included in most pipe stress programs. There is not much difference if the length of the beam is at least several times the cross sectional dimension. However, if the beam length is short the difference can be very great. Figure 4 shows a stack guided at a very short distance from the base to resist the wind. The problem is reduced to a fixed-supported beam applied with an end moment. As can be seen from the results tabulated, the shear deformation term has a very significant effect on the anchor and support loading if the guide is rigid.

For the curved pipe element, the formula given by J. E. Brock [5] can be used for cross check, if the cumbersom calculation can be managed. An alternative way is to divide the bend into multiple sections to see if the results agree with those of the undivided bend.

The curved pipe element involves flexibility and stress intensification factors. These factors are further influenced by the presense of flanged ends and internal pressure. In checking the intensification factor it is necessary to find out the program option in implementing the pressure effect. It is also desirable to understand the implications of the application. The pressure will tend to make the system more stiff, thus resulting in higher support loads against the thermal expansion. On the other hand it also tends to make the cross section more difficult to ovalize, thus reducing the stress intensification. The



Calculation Method	Support Fr (N)	Anchor Fa (N)	Anchor Ma (N-m)
With Shear Deform. Rigid Restraint	77,200	-53,700	63,400
Without Shear Deform Rigid Restraint	182,300	-158,800	-64,700
With Shear Deform. 17500 N/mm Rest.	1,800	21,600	155,300
Without Shear Deform 17500 N/mm Rest.	1,500	22,000	155,700

Figure 4, Effect of Shear Deformation Term

problem is that when the pressure is removed the temperature will likely to stay at near operating for some time. At this time the pipe moment is not reduced but the pressure is not there to help prevent ovalization. Therefore, the logical application is to take into account the increased stiffness, but not the decreased stress intensification.

SPECIAL FEATURES

Each pipe stress computer program has its own special features. These features are normally not available in benchmark problems. Their functions need to be checked by special schemes. Since it is not possible to cover all the features, this discussion will concentrate on three popular items. They are support friction, bellow elements, and thermal bowing.

(a) Support Friction

The support friction has a very significant effect on the analysis results in certain cases. The areas most sensitive to the friction are rotating equipment piping, long offsite piping, and transmission pipe lines. For instance, at a large rotating equipment, the friction due to a single support can often determine if the piping load exceeds the allowables or not.

There are different ways of implementing the friction effect in the program, but they are not all equal. Some methods require more computer time but are more inherently stable. Others are quick but prone to be unsatble. A detailed discussion on this subject is given in a separate topic [6]. In this paper the discussion is limited to the quick check of the results.

The validity of friction application depends on the type of the system analyzed. If the system is relatively rigid then the analysis tends to be correct regardless of which method is used. On the other hand, if the system is relatively flexible then the correct analysis can only be achieved with certain methods. This is because in a flexible system the friction not only affects pipe force, it also has the potential of changing the direction of the movement. To check the friction feature, it needs to check its application on a flexible system. With a support load report similar to Table 1 the function of the friction can be checked easily by the following steps:

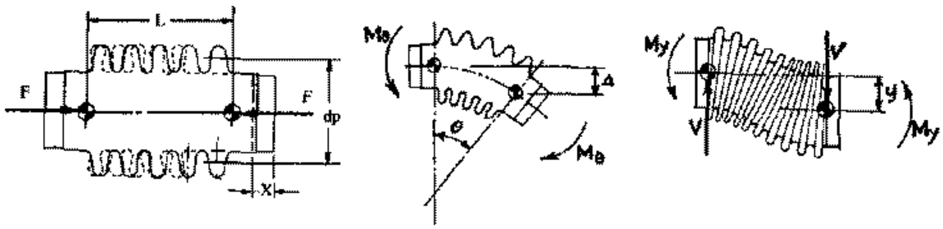
(1). If the piping is moving, then the resultant friction force should be equal to the normal support force multiplied by the friction factor. The direction, acting on the support, should be the same as the pipe movement. It reverses when acting on pipe.

(2). If the piping is stopped by the friction and not moving then the friction force should be equal or smaller than the full friction force calculated in (1).

(3). Most importantly, the above friction force is applied to the system. This can be checked by balancing the nodal forces at the support location. With a support load report similar to Table 1, the application of the friction can be checked by comparing the total friction force against the total system force. They should be the same if no other external force is applied to the system.

(b). Bellow Element

Bellow expansion joints can be simulated by the conventional zero length flexible connectors. However, to be able to represent the versatility of the bellow arrangements, the use of bellow elements is prefferred. With the bellow element, the program can easily simulate all the common bellow expansion joints such as single bellow, tied bellows, universal joints, and pressure balanced universal joints. The program will correctly apply the flexibility of the joint in all the translational and rotational directions. It also applies the proper pressure thrust force at the end of the bellow. The more advanced program can also combine all three dimensional motions to calculate the equivalent maximum axial displacement



(1) Axial Motion (2) Rotation (3) Lateral Motion

Figure 5, Elementary Function of Bellows

per convolution. This is the vital information used by the manufacturers to check the acceptability of their bellows.

Implementation of the bellow element involves some tricky maneuvers, but to check is simple. The Expansion Joint Manufacturers Association (EJMA) has a set of formulas [7] that can be used readily for checking the function of the bellow element. These formulas are copied below for easy reference.

$$ex = x / N$$

$$e\theta = \theta \cdot dp / (2N)$$

$$ey = 3\, dp \cdot y / (N \cdot L)$$

$$F = fw \cdot ex = (fw/N) \cdot x = Ka \cdot x$$

$$M\theta = fw \cdot dp \cdot e\theta / 4 = [(fw/N) \cdot dp^2 / 8] \cdot \theta$$

$$V = fw \cdot dp \cdot ey / (2L) = 1.5\, (fw/N)(dp/L)^2 \cdot y$$

$$My = fw \cdot dp \cdot ey / 4 = [0.75\, (fw/N) \cdot dp^2 / L] \cdot y$$

Where,

- ex = Axial displacement per convol. due to x
- $e\theta$ = Axial displacement per convol. due to θ
- ey = Axial displacement per convol. due to y
- x = Differential axial displ. across bellow
- θ = Differential rotation across bellow
- y = Differential lateral displ. across bellow
- N = Number of convolution
- dp = Pitch diameter of the convolution
- L = Effective length of the bellow element
- F = Axial force required to move x
- fw = Axial spring rate per convolution
- $M\theta$ = Moment required to bend θ
- V = Lateral force required to move y
- My = Moment created by y- movement
- Ka = Axial spring rate of the bellow element

From the above formulas it is clear if, for instance, the axial spring rate, pitch diameter, and the bellow length are given, then the spring constants in all the other directions can be determined. The equivalent axial displacement per convolution can also be found without needing additional data.

In checking the bellow function, a few items need to be further explained. As can be seen from the formula, the lateral spring rate is inversely proportional to the square of the bellow length. The axial deformation expected during operation can have a significant effect on the lateral spring rate. The analyst should try to input the shortest possible length in the analysis. It should also be noted that by laterally moving the bellow not only creates lateral force, but also the bending moment. In bending the bellow, the EJMA formula signifies that a lateral movement as well as a rotation is being created.

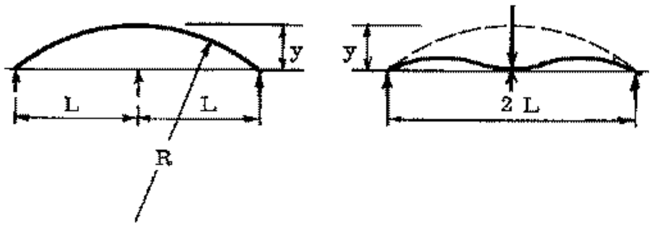
The checking can be done easily by fixing one end of the bellow element and apply the loading or displacement at the other end. This eliminates the trouble of finding the differential displacements of both ends. However, the real function of the bellow can only be evaluated by the checking of the differential movements. Once the elemental function is checked, its application to the piping assembly is not much different from the other elements.

(c). Thermal Bowing

Piping is normally assumed to have a uniform temperature across its cross section. However, due to stratified flow or some other reason, the temperature can vary greatly between the top and the bottom of the pipe. This situation can occur during the startup of large steam lines [8] or cryogenic lines [9]. It can also occur at a petrochemical transfer line hwen it is being quenched or when it has coke forming at the bottom of the pipe. When the temperature around the cross section is not uniform the pipe will form an arc shape. This bowing phenomenon may or may not create damaging stress in the pipe itself depending on the shape of the temperature distribution. If the distribution is linear, then no internal stress is created. If the distribution is not linear then large internal stress may be created. In either case, the bowing has the potential of creating huge displacements and rotations in the pipe. This huge movement can tear off connections if enough flexibility is not provided.

The bowing feature can be checked with two simple steps. Figure 6 shows a two span simply supported pipe. If no clamp or hold-down is installed in the mid-span, the pipe at mid-span will move up due to bowing. The amount of the move-up can be checked against the formula derived using the linear temperature distribution [9]. That is if the difference in expansion rate between the top and the bottom of the pipe is e mm/mm, the pipe diameter is d mm, then the radius of the curvature is $R = d/e$ mm. If the span length is L mm, then the expected move-up displacement $y = R - \sqrt{R^2 - L^2}$ mm.

After the bowing movement is checked, the (b) case can be used to check the combined effect. In this case, the mid-span is rigidly held down. The hold down load can be checked by using the simple beam formula applied with a concentrated force at the mid-span. The hold down load should equal the concentrated load with which a mid-span displacement of y mm is created. Of course the weight and other loads should not be included in making this check.



(a) Free Bowing (b) Hold Down Bowing

Figure 7, Bowing Function

STRESS REPORT

A well laid-out stress report can facilitate the checking of the analysis. The stress report should contain, in one continuous printout, all the input, interpretation of the input, generated system information, load case results, and stress and load compliance tables. Other graphical or tabular forms of presentations which are not integral parts of the output should be clearly identified for its association with the stress report.

The most important item to be checked on a stress report is the truthfulness of the mathematical model. This generally refers to the correctness of the input data, but includes also the correct interpretation of the program requirements. A good input echo and a good isometric picture generated directly from the input data can be very helpful.

(a). <u>Input Echo</u>

With the popularity of the menu input approach, the input data made by an analyst is converted into another way of expression almost immediately. The so called input echo printed out by some programs can not even be recognized by the analyst who have entered the input in the first place. It is, needless to say, a nightmare to the checker. The input echo which is the most important document of the report should be readable not only to the machine, but also to both the analyst and checker. The best form is the one which preserves all the styles and letters the analyst has entered. Figure 7 shows the echo of the input which is done with the popular piping language.

(b). Faithful Isometric

A good isometric is an invaluable tool for quick checking the mathematical model. The isometric need not be pretty but has to be faithful. It should show all the bends, valves, flanges, and other components. The restraints should be shown in the correct location and also in the correct direction. The nodal numbers should all be identified properly. Figure 8 shows the typical isometric drawing which is printed directly and automatically from the input data.

REFERENCES

1. The M.W. Kellogg Company, "<u>Design of Piping Systems</u>," 2nd edition, John Wiley, 1964.

2. Tuba, T.S., and Wright, W.B. editor, "Pressure Vessel and Piping, 1972 Computer Programs Verification," ASME, 1972.

3. Bezler, P., Hartzman, M., and Reich, M., "Piping Benchmark Problems," Brookhaven National Laboratory, NUREG/CR-1677, August, 1980.

4. "SIMFLEX-II User's Manual," rev-d, 1987, Peng Engineering, PO Box 801167, Houston, TX 77280

5. Brock, J.E., "Expansion and Flexibility", Chapter 4, <u>Piping Handbook</u>, 5th edition, edited by King and Crocker, McGraw-Hill, 1967.

6. Peng, L.C., "treatment of Support Friction in Pipe Stress Analysis," ASME/JSME PVP Conference, Hawaii, July 23-27, 1989.

7. EJMA, "Standards of the Expansion Joint Manufacturers Association, Inc.", 707 Westchester Ave, White Plains, NY 10604

8. Machacek, S., and Zelenka, T., "Design and Operation of Large Diameter (1.67m) steamline," ASME/CSME PVP Conf, Montreal, June 1978.

9. Flieder, W.G., Loria, J.C. and Smith, W.J. "Bowing of Cryogenic Pipelines," Trans ASME, J. Appl Mech, Sep. 1961, pp 409-416.

Figure 7, Input Data Echo

```
50, EL=34.44, TEE
60, EL=33.0, BR
65, Z=-1.22, LSX(-3, 3), NLY(,,0.4,-1)
68, Z=-1, WNF, +FGTV, +WNF
70, Z=-5, STX, NLY(,,0.4,-1)
90, Z=-6, LSX(-6, 0), NLY(,,0.4,-1)
95, Z=-2.0, BR
100, X=0.61, LSZ-6
105, X=2.6, SPRING=2
```

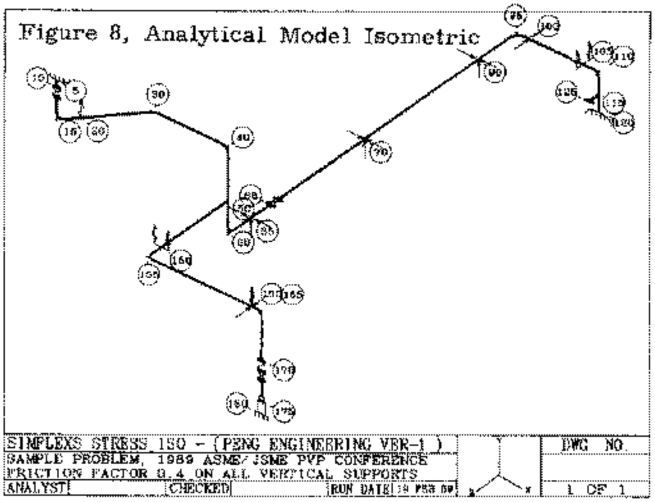


Figure 8, Analytical Model Isometric